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VORTEX TUBE COOLING (U)

FINAL REPORT

Project 11061101A91A-00-057 EF

SEPTEMBER 1971

by

ROBERT B. SHERFY
ENVIRONMENTAL EQUIPMENT DIVISION
MECHANICAL TECHNOLOGY DEPARTMENT

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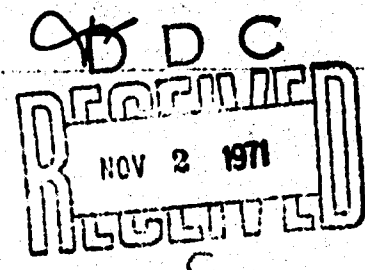


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Details of illustrations in
this document may be better
studied on microfiche

IN-HOUSE LABORATORY INDEPENDENT RESEARCH
FINAL REPORT
17 September 1971

- I. Title: Vortex Tube Cooling
- II. Project Director: Robert B. Sherfy
(Previously J. Palmer Dauphin and Oscar Oldberg.)
- III. Objective: Investigate the operation and efficiency of the Vortex Tube, the possibility of improvement in efficiency and the feasibility of military applications.
- IV. Background: A brief history and description of the Vortex Tube is located in appendix III.
- V. Accomplishment during the Program:

A. FY 1970: The following accomplishments occurred during FY 1970.

1. A literary search was made for articles on books relating to the Vortex Tube and its use. The list of 75 publications is presented in appendix IV to this report. Also the Russian book described in the next paragraph contains a reference list of 145 writings which complements this list.

2. The book "The Vortex Effect and It's Application in Technology" by A. P. Merkulov, 1969 (in Russian) was translated.

3. A commercial 25 cfm tube was fitted with cooling fins on the hot end and ambient air was circulated past these fins. The additional cooling of the hot side produced a 10% increase in cooling capacity at the cold end.

4. Large Vortex Tubes were purchased for future low pressure testing.

B. FY 1971: The following accomplishments occurred during FY 1971 in the Vortex Tube Program.

1. Low pressure operational tests were preformed on a large commercial Vortex Tube. The main purpose was to determine if a "worthwhile" increase in coefficient of performance (COP) occurred at low inlet pressures. As described in the test report, appendix I, the COP increased approximately from 0.08 to 0.12. See a further discussion of these results in Section VI, conclusions.

2. A cursory review was made of the Russian book "The Vortex effect and It's Application in Technology ", 1969 by A. P. Merkulov.

In brief, it offers little new material but discusses at length and at a technical level the Vortex effect, thermodynamic analysis, practical applications and analyzes practical means of increasing the thermodynamic effectiveness of the tube. Good coverage is given to aircraft and industrial uses of the tube.

- VI. Conclusions: The Vortex Tube has been the target of intensive investigation since its invention in the 1930's. Improvements have brought it to its present level of perfection and specific uses in industry have made it a commercial off-the-shelf item. It's main uses are for spot cooling such as for a cooled suit for fire fighters. The Vortex Tube's major advantages are small size and weight, low cost, high reliability and low maintenance. It's major disadvantage is it's poor efficiency or low COP when considered with a compressor needed to supply compressed air. As mentioned in appendix I, the COP of the Vortex Tube - Air Compressor system is only in the 0.05 to 0.12 range compared to, perhaps, 0.8 to 1.0 for thermoelectric and closed air cycle units and 1.0 to 1.4 for military vapor cycle air conditioners.

With all of the work having been done on the Vortex Tube since its inception, it is, perhaps, unusual that an exact explanation of its action has not been proved. There are many theories of its operation; some more plausible than others but none having been proven totally correct. Obviously the action within the Vortex Tube is a complex gas-dynamic process occurring in a turbulent stream of viscous compressed gas. It is sufficiently complicated to defy, thus far, attempts at complete analytical solution to the problem.

Vortex Tubes are now developed and effects are predicted through the use of semi empirical formulae. The known relationships which are most important in considering the Vortex Tube for military cooling use are as follows:

1. The total mass flow (required inlet air flow) is proportional to the absolute inlet pressure (PSIA). (Cold side discharges to atmosphere.)

2. Most efficient operation appears to be between $\mu = .6$ and $\mu = .7$ for most tubes. μ = Cold mass fraction = $\frac{\text{cold side SCFM}}{\text{inlet SCFM}}$

3. Temperature relationships are approximately as follows for ideal conditions with dehumidified air: $\mu (T_1 - T_c - JT) = (100 - \mu) (T_h - T_1 + JT)$ where: T_1 is inlet air temp ($^{\circ}\text{F}$)

c is cold

h is hot

JT is Joule - Thompson effect which can be found in thermodynamic tables but is roughly 4°F per 100 psig pressure drop.

The poor efficiency of the Vortex Tube - compressor system dictates that it be used only where the power penalty is not important or where the specific use of the tube allows a much smaller amount of cooling to accomplish the required purpose. These conditions coupled with the tube's trouble free operation would make the Vortex Tube an attractive solution.

Considering only the cost of furnishing power for air conditioning, we find that if it takes 5 tons of cooling to air condition the crew compartment of the Main Battle Tank, about 1/2 ton of cooling would probably cool the crew using cooled seats and gaspers (nozzles directed at the person such as the adjustable air supply outlet in the overhead console on commercial aircraft.) Now, if the vehicle had a readily available source of compressed air, such as a compressor used also for other purposes, or bleed air from a gas turbine, both the 6,000 BTUH Vortex Tube and a 60,000BTUH vapor cycle would need the same power input to accomplish roughly the same purpose, and therefore the Vortex Tube would be the preferred solution by virtue of its small size, weight, cost and high reliability. Under any other imaginable situation, the Vortex Tube would be overlooked because of its poor efficiency.

VII. Recommendations:

Since only special circumstances will justify the use of Vortex Tube cooling, these special situations should be studied. This may uncover the means whereby much needed cooling can be made practical for fighting vehicles.

It is therefore recommended that no more time be spent on attempts to improve Vortex tube efficiency, but that efforts be directed toward the application of commercial Vortex Tubes for use with cooled seats, gaspers, and other personal cooling and spot cooling devices.

FY 1971 Test Data

LOW PRESSURE VORTEX TUBE OPERATION

PROJECT DIRECTOR: Robert B. Sherfy

TEST TECHNICIAN: Ralph E. Beahm

JUNE 1971 TESTS

ENVIRONMENTAL EQUIPMENT DIVISION
MECHANICAL TECHNOLOGY DEPARTMENT
U.S. ARMY MOBILITY EQUIPMENT RESEARCH AND DEVELOPMENT CENTER
FORT BELVOIR, VIRGINIA 22060

APPENDIX I

IN-HOUSE LABORATORY INDEPENDENT
RESEARCH
TEST REPORT

September 15, 1971

- I. TITLE: Low pressure vortex tube operation.
- II. PROJECT DIRECTOR: Robert B. Sherfy
- III. OBJECTIVE: Study the operation of vortex tube at low pressures to determine any advantages over high pressure operation.
- IV. ACCOMPLISHMENT:

A. Discussion: Tests were conducted at the USAMERDC, Fort Belvoir, Va. air conditioning laboratory test facility during June of 1971. The test item was a nominal 250 cfm commercial vortex tube, model 454-250, manufactured by Vortec Corporation of Cincinnati, Ohio (Formerly Fulton Cryogenics, Inc.) The test item is depicted in Figure 5, and the test set up is shown in Figures 6, 7 and 8. Tests were conducted at 10, 15, 20, 25 and 30 psig inlet pressure with a few readings taken at 80 psig for comparison. It was found that the "building" air supply was dry enough so that there was never any latent cooling involved in these vortex tube tests.

As with all vortex tubes the amount of cooling air-passing from the cold side is controlled by a valve which restricts the issuance of air from the hot end. At each inlet pressure, readings were taken at various hot valve settings. Enough settings were used to obtain data describing the temperature drop and cooling capacity curves shown respectively in Figures 1 and 2 except that some data deficiency occurred where code tester nozzles accuracy ranges were exceeded.

Other possible inaccuracies resulted from heat losses through insulation and direct conduction between hot and cold portions of the vortex tube. However, this test was intended to indicate results of a practical application of the tube, rather than an "ideal" laboratory setup and therefore served the required purpose. The resulting curves can be compared with those of Vortec "advertised" data as shown in Figure 3 for 20 and 100 psig inlet pressures. It is suspected, that the Vortec Bulletin air temperatures were sensed at a point inside the tube which would yield a cold temperature lower than that which might be available for practical use in an actual cooling application.

Again, in order to indicate operation under "practical" conditions, the coefficient of performance (C.O.P.) was calculated using actual compressor power requirements, rather than theoretical air horsepower factor. The resulting COP's, calculated in BTUH of cooling/BTUH equivalent of the required actual compressor input, used the high points from the capacity curves. COP values from test data are presented in Table 3 and from Vortec "ideal" data (calculated on the same practical basis) in Table 5.

These COP values were plotted on Figure 4. The MERDC test data was inconsistent and inconclusive, mainly because of the attempt toward a "practical application". The "ideal" curve, however, indicates the trend toward a higher COP at lower inlet pressures.

We then find that optimum COP values range from 0.044 to 0.63 for test data and from 0.083 to 0.121 for the "ideal" data. The ideal data indicates an increase in COP of only 0.04 for a decrease in inlet pressure from 100 to 20 psig.

B. Conclusion: For a decrease in inlet pressure from 100 to 20 psig, there is only an increase in COP of 0.04. This small increase is applicable to an already very low COP. For instance, the vortex tube COP of 0.044 to 0.12 looks unattractive compared to, say, 0.8 to 1.0 for Thermoelectric and closed air cycle air conditioners and 1.0 to 1.4 for conventional vapor cycle units. Also, as inlet air pressure is decreased, the vortex tube must increase in physical size for the same BTUH capacity.

At the same time, it must be remembered that the vortex tube itself is relatively small in volume, is low in first cost, and is practically maintenance free. It is when the source of compressed air is considered, with its' inherent efficiency, size, weight, cost and maintenance requirements, that the vortex tube appears the worst. If a "free" source of compressed air is available, the vortex tube becomes ideal for many purposes of spot cooling.

SYMBOLS AND ABBREVIATIONS

1. BHP - Brake horsepower.
2. BTUH - British Thermal Units per hour.
3. COP - Coefficient of performance.
4. C_p - Specific heat at constant press. (BTU/lb-°F).
5. #DA - Pound of dry air.
6. DB - Dry bulb temperature °F.
7. DP - Dew Point °F
8. GR - Grains of moisture
9. "HgA - Inches of Mercury pressure absolute.
10. "H₂O - Inches of water pressure head.
11. M_c - Mass Air flow thru cold end (lb/min).
12. M_h - Mass Air flow thru hot end (lb/min).
13. M_{in} - Mass Air flow in (lb/min).
14. P - Pressure PSIG.
15. PSIA - Pounds per square inch absolute.
16. PSIG - Pounds per square inch guage.
17. Q_c - Cooling capacity (BTUH).
18. SCFM - Standard cubic feet per minute air flow.
19. T - Temperature °F (Dry bulb unless otherwise noted)
20. Δt - Change in temperature (°F).
21. WB - Wet bulb temperature °F.
22. W_k in - Work input (in units as indicated).
23. $\frac{M_c}{M_{in}}$ - Cold Air Fraction = $\frac{M_c}{M_{in}}$

INSTRUMENTATION

1. Inlet Temperature - Type T Thermocouple, copper - constantan.
See also Item 2.
2. Potentiometer, No./OB.8, Honeywell model 15501836-01-01-2,
range - 100° to + 300°F, scale 359317-213, 120V, 60Hz, S/N
Y6803541001.
3. Inlet Dew Point Meter, No. AJ7, EG&G Mfg. Co., model 880,
S/N 260.
4. Inlet Pressure Gauge, USG, 0-200 psi, 1 lb. graduations.
5. Cold Side Dry Bulb Temperature, Brooklyn Thermometer Co.,
Thermometer No. 9B.3, length 16", range 30° to 122°F, gradua-
tions 2/10°F, S/NT7610.
6. Cold Side Wet Bulb Temperature, BTC Thermometer No. 9B.12,
length 16", range 14° to 122°F, graduations 2/10°F, S/N F5763.
7. Cold Side Air Flow Measurement, ASHRAE Code Tester 17A.1. Used
Nozzles 0.988" dia, factor 5.763; 2.0", factor 23.5; 3.0",
factor 53.0.
8. Cold Side Flow Manometer No. 11B.4, inclined model 40HE35,
S/WT63742, range 6", graduations 1/100", Mariam Instrument Co.
9. Cold Side Static Pressure Manometer, inclined, range - 0.20 to
+ 2", graduations 1/100", F. W. Dwyer Co., No. 11D.2.
10. Hot Side Dry Bulb Temperature, Type T Thermocouple, Copper-constantan.
See also item 2.
11. Hot Side Air Flow Measurement, ASHRAE Code Tester 17A.9. Nozzles
used were 0.7517" dia, factor 3.243; 0.998", factor 5.763; 1.400",
factor 17.35.
12. Hot Side Flow Manometer, No. 11B.13, inclined, model 40 HE35, S/N
T45957, range 4", graduations 1/100", Mariam Instrument Co.
13. Hot Side Static Pressure Manometer, inclined, No. 11D.1, range -
0.20 to 2", graduations 1/100", F. W. Dyer Co.

VORTEX TUBE CALCULATIONS

COLD SIDE AIR FLOW AT 10 PSIG TUBE INLET PRESSURE.

* = PREDETERMINED CONSTANTS.

- ① AIR IN TEMP DB = 67.2 °F
- ② " " " WB = 50.6 °F
- ③ GR. MOIST / # DA (PSYCH. CHART) = 27 GR / #
- ④ STATIC PRESS. AFTER NOZ = -4.02 "H₂O
- ⑤ ΔP AT NOZZLE = 4.02 "H₂O
- ⑥ STATIC PRESS. BEFORE NOZ. "H₂O
= ④ + ⑤ = -4.02 + 4.02 = 0 "H₂O
- ⑦ BAROMETER (WEATHER BUR) = 29.91 "Hg A
- ⑧ CORR. BAR = ⑦ - 0.14* = 29.91 - 0.14
= 29.77 "Hg A
- ⑨ STATIC P2. BEFORE NOZ "Hg A
= ⑧ + $\frac{⑥}{13.62*}$ = 29.77 + $\frac{0}{13.62}$
= 29.77 "Hg A
- ⑩ AIR TEMP AT NOZ. DB. = 67.2 °F
- ⑪ MOISTURE CORR.
= $1.6055 * \frac{(③ + 4360*)}{(③ + 7000*)}$
= $1.6055 \frac{(27 + 4360)}{(27 + 7000)} = 1.0023$

⑫ SPECIFIC VOLUME @ NOZZLE $\text{FT}^3/\#$ MIX.

$$V_n = \frac{.754^* (\textcircled{10} + 460^*) \times \textcircled{11}}{\textcircled{9}}$$

$$= \frac{.754 (67.2 + 460) \times 1.0023}{29.77}$$

$$= \underline{13.384 \text{ FT}^3/\# \text{ AIR-VAPOR MIXTURE}}$$

⑬ NOZZLE FACTOR = 5.763 *

⑭ AIRFLOW $\#/\text{MIN}$ = $\textcircled{13} \sqrt{\frac{\textcircled{5}}{\textcircled{12}}}$

$$= 5.763 \sqrt{\frac{4.02}{13.384}} = \underline{3.158 \#/\text{MIN.}}$$

⑮ SPEC. VOL. @ INLET $\text{FT}^3/\#$ MIXTURE

$$V_{in} = \frac{.754 (\textcircled{1} + 460) \times \textcircled{11}}{\textcircled{8}}$$

$$= \frac{.754 (67.2 + 460) \times 1.0023}{29.77}$$

$$= \underline{13.384 \text{ FT}^3/\# \text{ AIR-VAPOR MIX.}}$$

⑯ AIRFLOW CFM @ NOZ = $\textcircled{14} \times \textcircled{12}$

$$= 3.158 \times 13.384 = \underline{42.27 \text{ CFM}}$$

CALCULATIONS

VORTEX TUBE

$$PSIA = PSIG + 14.7$$

$$\text{Ratio of compression} = \frac{\text{Inlet psia}}{\text{Atmospheric psia}}$$

$$SCFM = \text{Flow in pounds per min.} \times 13.35 \text{ (SCFM per pound)}$$

$$\text{Required BHP} = \frac{(\text{BHP}/695 \text{ cfm}) \times SCFM}{695}$$

TABLE 1

APPROXIMATE REQUIRED COMPRESSOR POWER

PSIG	10	15	20	25	30	40	60	80	100
PSIA	24.7	29.7	34.7	39.7	44.7	54.7	74.7	94.7	114.7
RATIO OF COMPR'N	1.68	2.02	2.36	2.70	3.04	3.73	5.08	6.44	7.80
BHP/695 CFM *	35	45	54	62.5	70	85	110	134	-
MERDC TEST SCFM	52	67	79	94	107	-	-	-	-
THEORETICAL SCFM **	-	-	-	-	-	131	179	227	275
TOTAL FLOW #/MIN ***	3.91	5.0	5.88	7.0	8.0	9.8	13.4	17.0	20.6
REQUIRED BHP ****	2.6	4.3	6.1	8.4	10.8	16.0	28.3	43.7	61.5
KBTUH EQUIV.	7.8	12.9	18.3	25.2	32.4	48.0	85.0	131	185

* From 1958 Marks Handbook, 6th edition, page 14-52, fig. 28,

1,000,000 cu. ft./day = 695 cfm

** Theoretical scfm not derived from MERDC tests but calculated using the relationship that total vortex tube flow rate is proportional to absolute pressure and is based on data at 30 psig.

*** Flows are average test values.

**** Extrapolated from Marks Handbook (See *)

***** Btu equivalent = $\frac{\text{BHP} \times 2547}{0.85} = 3000 \times \text{BHP}$

where motor eff. is assumed to be 85%.

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3 Sept 71

$$Q_c = m_{in} \times \mu \times C_p \times 60 \times \Delta t$$

$$\text{and since: } m_{in} \times \mu = m_c$$

$$Q_c = 0.241 \times 60 \times m_c \times \Delta t$$

$$= 14.46 \times m_c \times \Delta t$$

TABLE 2 COOLING CAPACITY

Psig	μ	m_c^*	Δt	Q_c	Psig	μ	m_c^*	Δt	Q_c
10	.817	3.16	7.8	356	25	.850	6.03	12.2	1064
	.639	2.48	8.2	294		.679	4.78	15.8	1092
	.453	1.79	6.1	158		.388	2.69	23.0	875
	.320	1.28	7.9	146		.298	2.07	23.0	688
15	1.0	4.91	4.8	341	30	.248	1.74	22.9	576
	.727	4.10	10	593		.202	1.41	22.6	461
	.653	3.19	14.5	669		.173	1.22	22.2	392
	.48	2.37	13.0	446		1.0	7.81	2.2	248
20	.349	1.68	12.5	304	80	.849	6.73	19.2	1868
	.252	1.23	12.1	215		.692	5.41	25.7	2010
	1.0	6.09	7.9	696		.386	3.01	28.4	1236
	.847	5.10	11.4	841		.297	2.33	26.4	889
25	.770	3.96	15.8	905	AI-9	.252	1.98	24.0	687
	.488	2.92	19.0	802		.204	1.59	27.5	632
	.363	2.16	18.6	581		.338	3.27	26.0	1237
	.281	1.67	18.2	439		.165	1.28	25.9	479
30	.222	1.32	18.0	344	AI-9	.154	1.20	26.0	451
	.180	1.08	17.3	270		1.0	19.3	2.4	670
	1.0	7.03	3.6	366		.858	16.75	25.0	6055
	* MEASURED VALUES					.712	14.32	41.9	8676

TABLE 3

OPTIMUM COP ($Q_c/W_{k in}$)

INLET PRESSURE (PSIG)	OPTIMUM COOLING* Q_c (BTUH)	WORK IN** $W_{k in}$ (BTUH)	COP ($Q_c/W_{k in}$)
10	380	7800	.049
15	615	12900	.048
20	935	18300	.051
25	1110	25200	.044
30	2030	32400	.063

* CURVE PEAKS FROM FIG. 2

** FROM TABLE 1

Result: No significant trend in COP vs. inlet pressure can be determined from these test results.

Try using Theoretical data from commercial Vortex tube charts.
See next page.

AI-10

USAMERDC - ILIR

VORTEX TUBE TEST

1 SEPT 71
SHERFY

VORTEC MODEL 454+250

COLD SIDE TEMP. DROP
-VS-
COLD MASS FRACTION

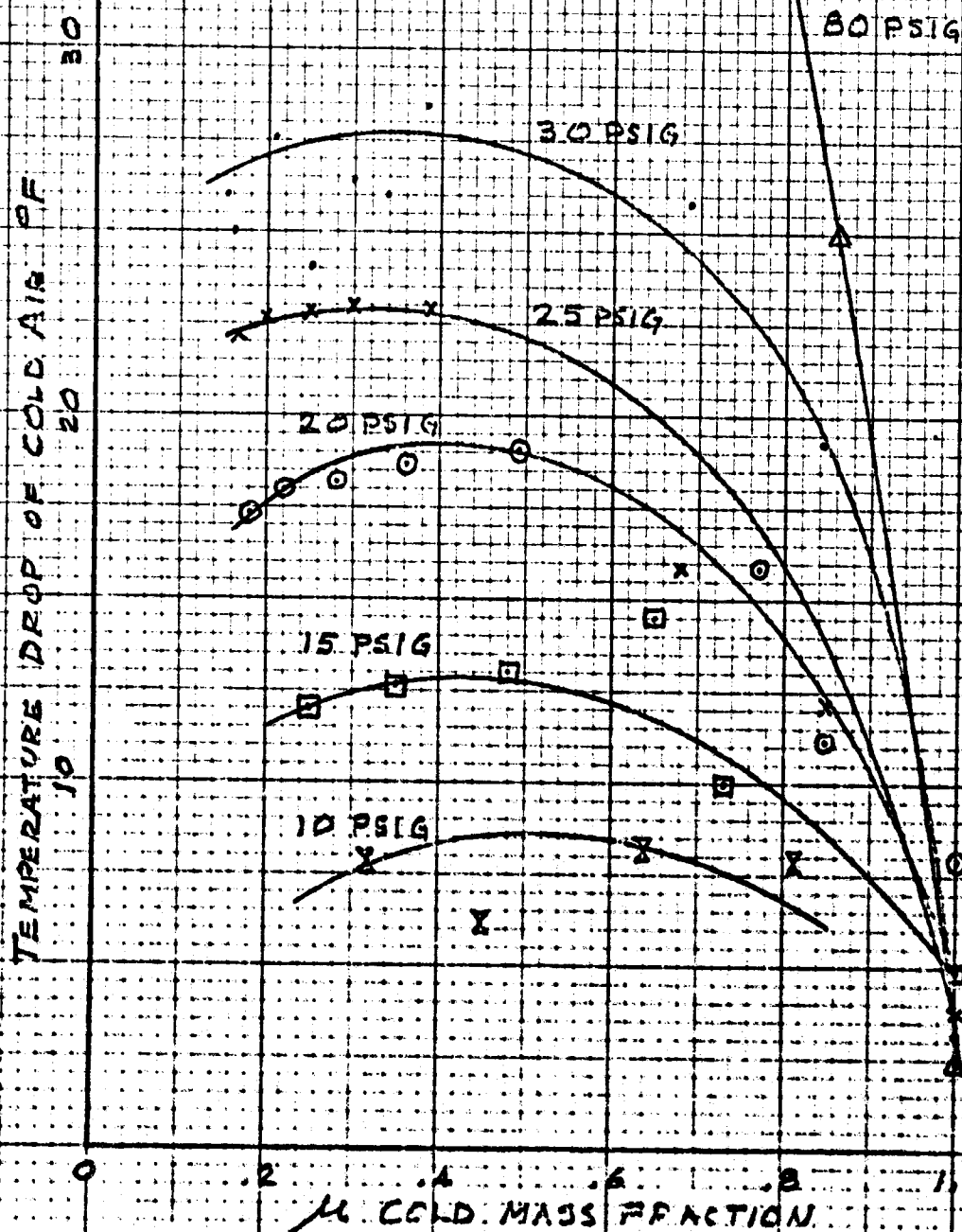


FIG. 1

A1-11

USAMERDC - ILIR VORTEX TUBE TEST

7 SEPT 71
SHERBY

VORTEC MODEL 454-250

BTUH COOLING
- VS -
COLD MASS FRACTION

80 PSIG
(USE "B" SCALE)

25 PSIG

20 PSIG

15 PSIG

10 PSIG
(USE "B" SCALE)

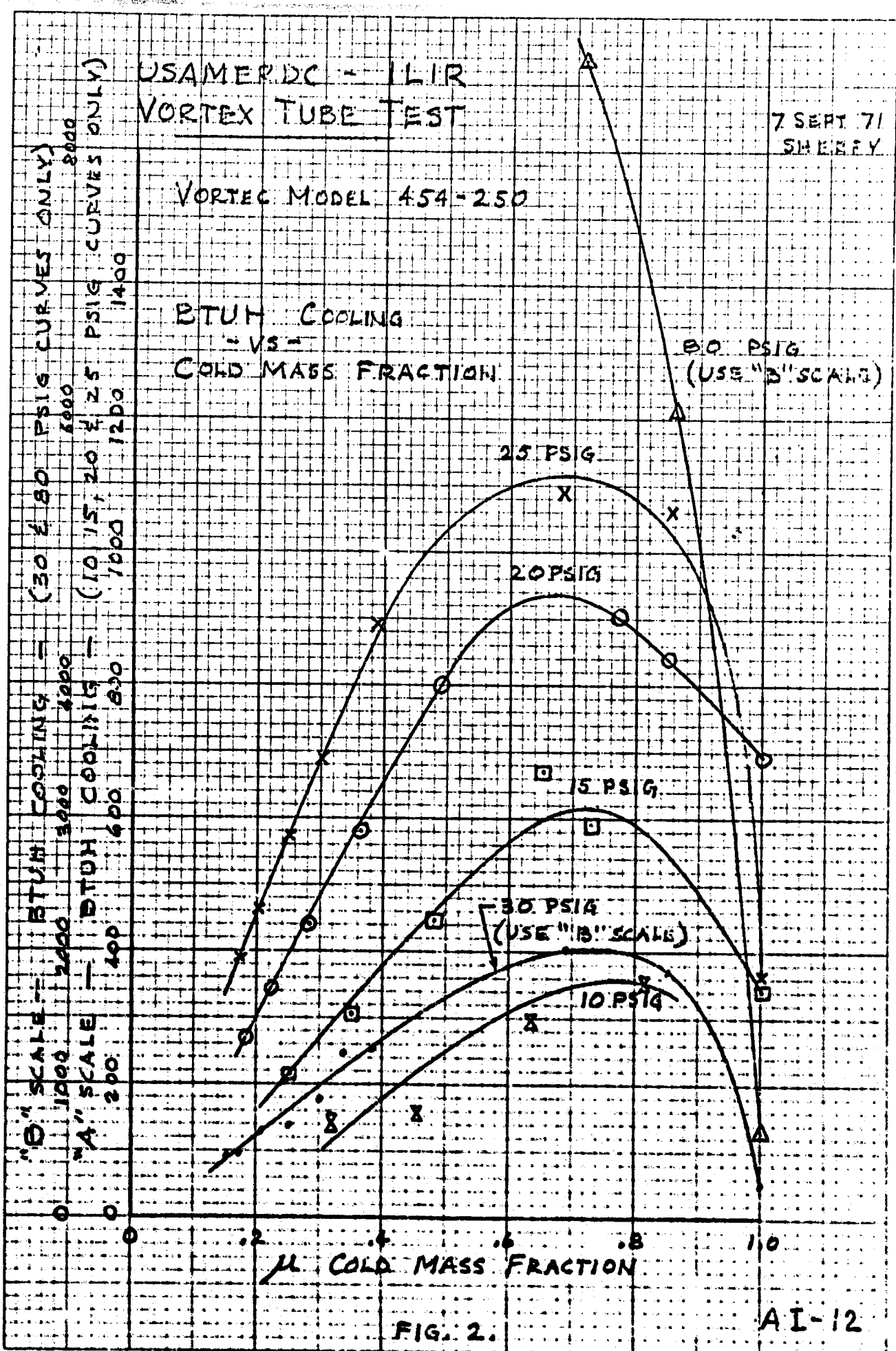
10 PSIG

μ COLD MASS FRACTION

FIG. 2.

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THEORETICAL CALCULATIONS

Reference : Fulton Cryogenics (now Vortec Corp.) Bulletin No. 2 - Rev., Mar. '66
Back page chart "Temperature Changes Produced by Best Large Vortex tubes".

$$Q_c = \mu \times \dot{m}_{in} \times c_p \times \Delta t \times 60$$

At 20 psig using Total flow from table 1, with $\mu = 0.10$,
 $\Delta t = 62.5^\circ$ from above ref.

$$\begin{aligned} \text{Then } Q_c &= 0.1 \times 5.88 \times 0.241 \times 62.5^\circ \times 60 \\ &= 531.4 \quad \text{BTU/h} \end{aligned}$$

See Table 4 for remainder of Q_c values for inlet pressures up to 100 psig.

TABLE 4 THEORETICAL COOLING CAPACITY

psig	m_{in}^*	μ	m_c	Δt	Q_c
20	5.88	0.0	0	63.0	0
		.1	.588	62.5	531
		.2	1.18	61.5	1056
		.3	1.76	59.5	1518
		.4	2.35	55.5	1894
		.5	2.94	50.5	2140
		.6	3.53	43.5	2212
		.7	4.12	36.0	2140
		.8	4.70	27.5	1865
		.9	5.29	17.0	1301
		1.0	5.88	0.5	43
40	9.8	0.0	0	92.0	0
		.1	.98	91.0	1290
		.2	1.96	88.0	2494
		.3	2.94	85.0	3614
		.4	3.92	80.0	4535
		.5	4.90	73.0	5172
		.6	5.88	62.5	5314
		.7	6.86	51.5	5109
		.8	7.84	38.0	4308
		.9	8.82	25.5	3252
		1.0	9.8	1.0	142

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* From table 1

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TABLE 4 CONT'D

psig	min *	μ	m_c	Δt	Q_c
60	13.4	0.0	0	108	0
		.1	1.34	107	2074
		.2	2.68	104	4030
		.3	4.02	100	5813
		.4	5.36	93	7208
		.5	6.70	84	8138
		.6	8.04	73	8787
		.7	9.38	59.5	8070
		.8	10.72	44.5	6897
		.9	12.06	28.5	4970
		1.0	13.4	2.0	383
80	17.0	0.0	0	120	0
		.1	1.70	119	293
		.2	3.40	115	5654
		.3	5.1	110	8112
		.4	6.8	102	10029
		.5	8.5	92	11308
		.6	10.2	80	11799
		.7	11.9	65.5	11272
		.8	13.2	49.6	9636
		.9	15.3	31.0	6858
		1.0	17.0	3.0	737

TABLE 4 CONT'D

P_{sig}	W_{in}^*	u	W_c	Δt	Q_c
100	20.6	0.0	0	128	0
		.1	2.06	127	3783
		.2	4.12	123	7328
		.3	6.17	118	10544
		.4	8.24	110	13107
		.5	10.30	99	14745
		.6	12.36	86	15371
		.7	14.42	70.5	14700
		.8	16.48	53.0	12692
		.9	18.54	33.0	8847
		1.0	20.6	4.0	1192

THEORETICAL

TABLE 5 OPTIMUM COP (Q_c/WK_{in})

INLET PRESSURE (PSIG)	OPTIMUM COOLING* Q_c (BTUH)	WORK IN WK_{in} ** (BTUH)	COP (Q_c/WK_{in})
20	2212	18300	0.121
40	5314	48000	0.111
60	8487	85000	0.100
80	11799	131000	0.089
100	15371	185000	0.083

* CURVE PEAKS AT $\mu = 0.6$.

** FROM TABLE 1.

RESULT : COP does appear to increase at lower inlet pressures. However for a decrease in inlet pressure from 100 to 20 psig, the COP increased only about 0.04 or 4%.

USAMERDC - ILIR VORTEX TUBE PROGRAM

COOLING CAPACITY AND TEMP. DROP VS COLD MASS FRACTION

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SHERREY

DATA FROM
FULTON CRYOG.
INC. (NOW
VORTEC CORP.)
BULLETIN 2-REV
MARCH 1966
BACK PAGE CHART
SEE ALSO TAB.
4, THIS REPORT.

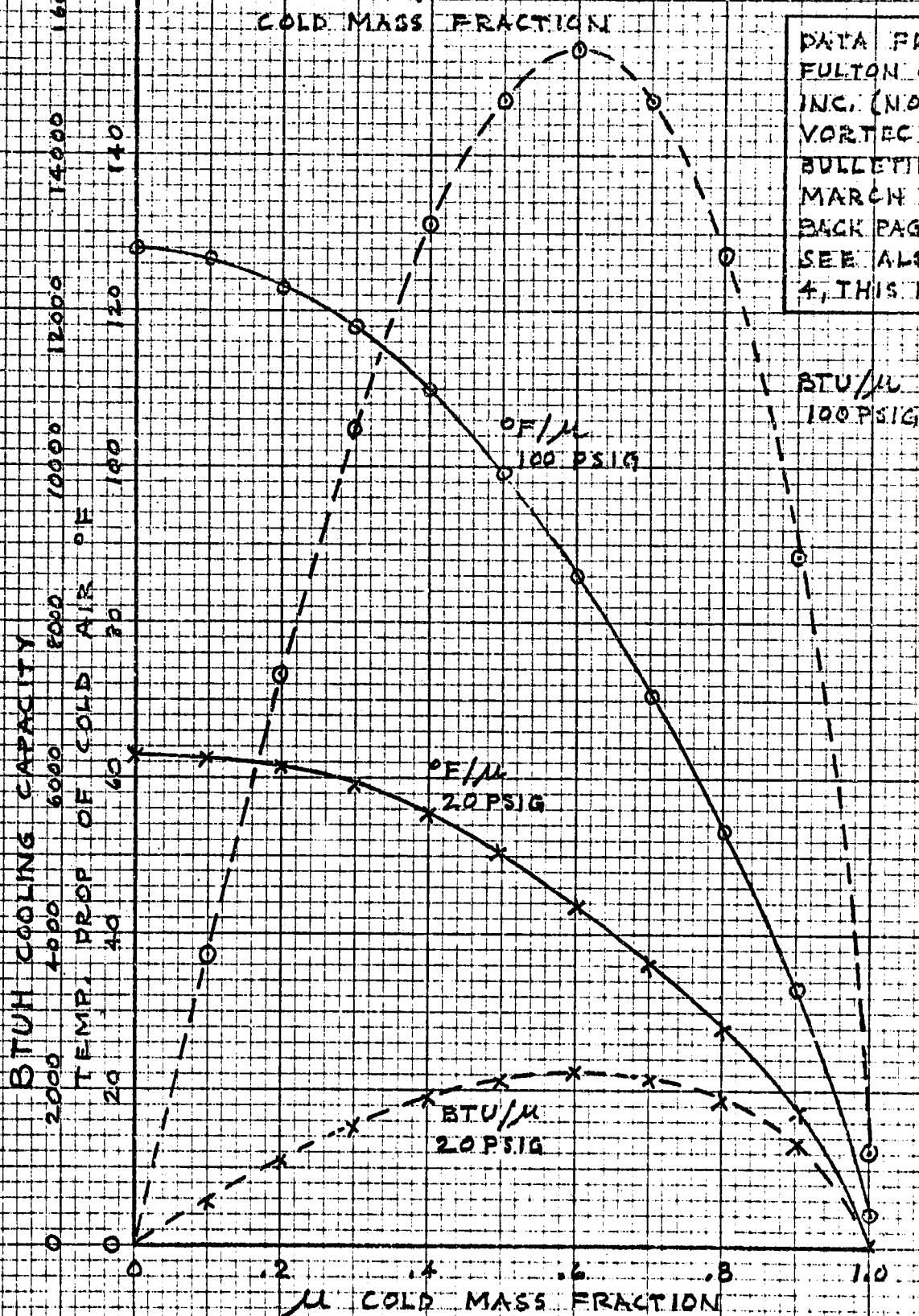


FIG. 3

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USAMERDC - ILIR

VORTEX TUBE PROGRAM

9 SEPT 71
SHERLY

COEFFICIENT OF PERFORMANCE
VS
INLET AIR PRESSURE

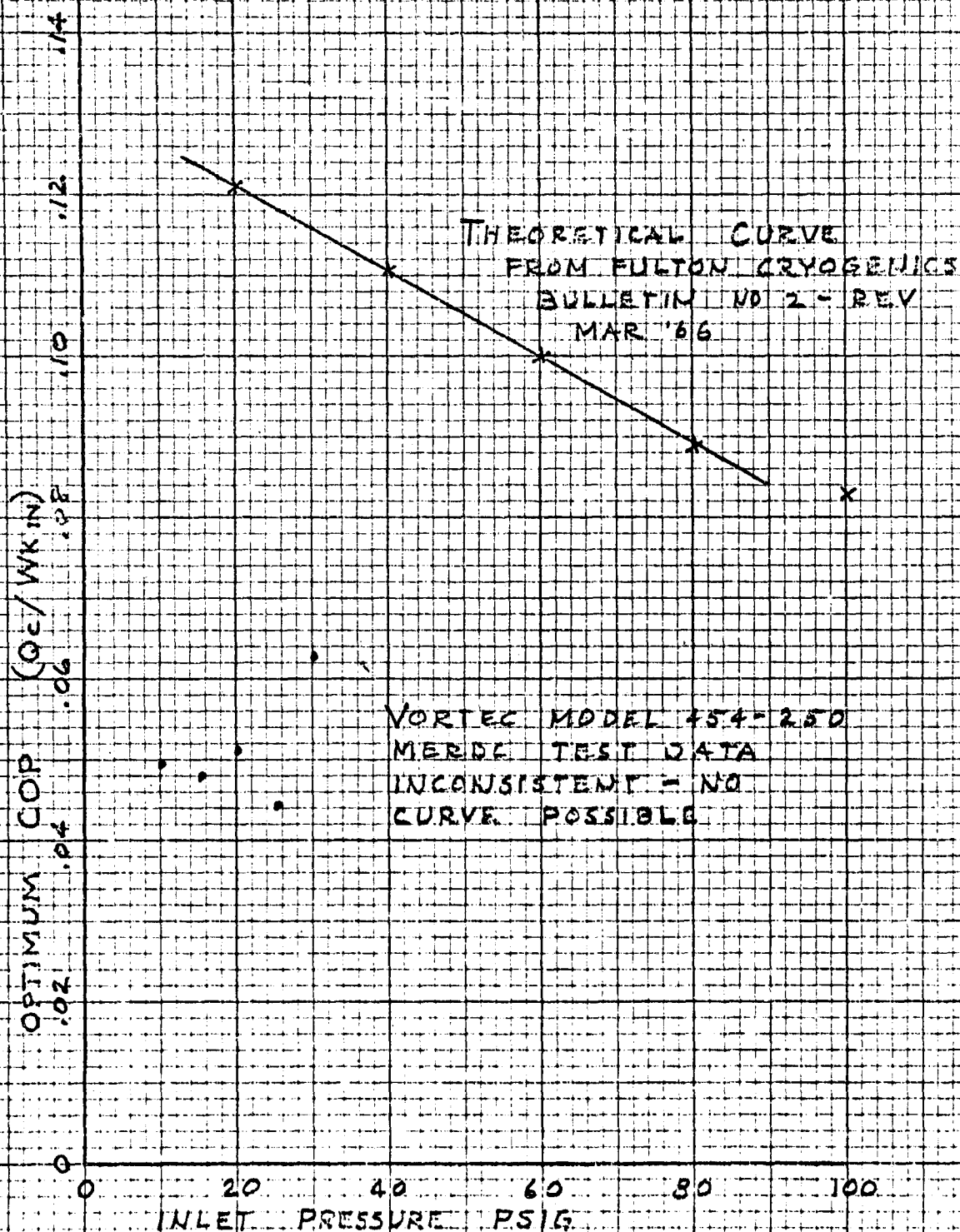


FIG. 4

AI-19

VORTEC CORPORATION

VORTEX TUBE MODEL NO. 454 - 250

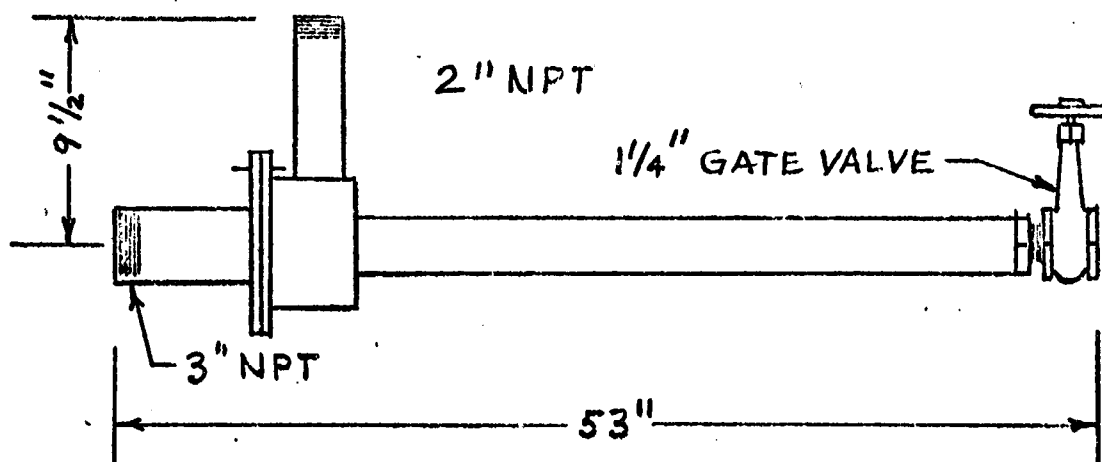
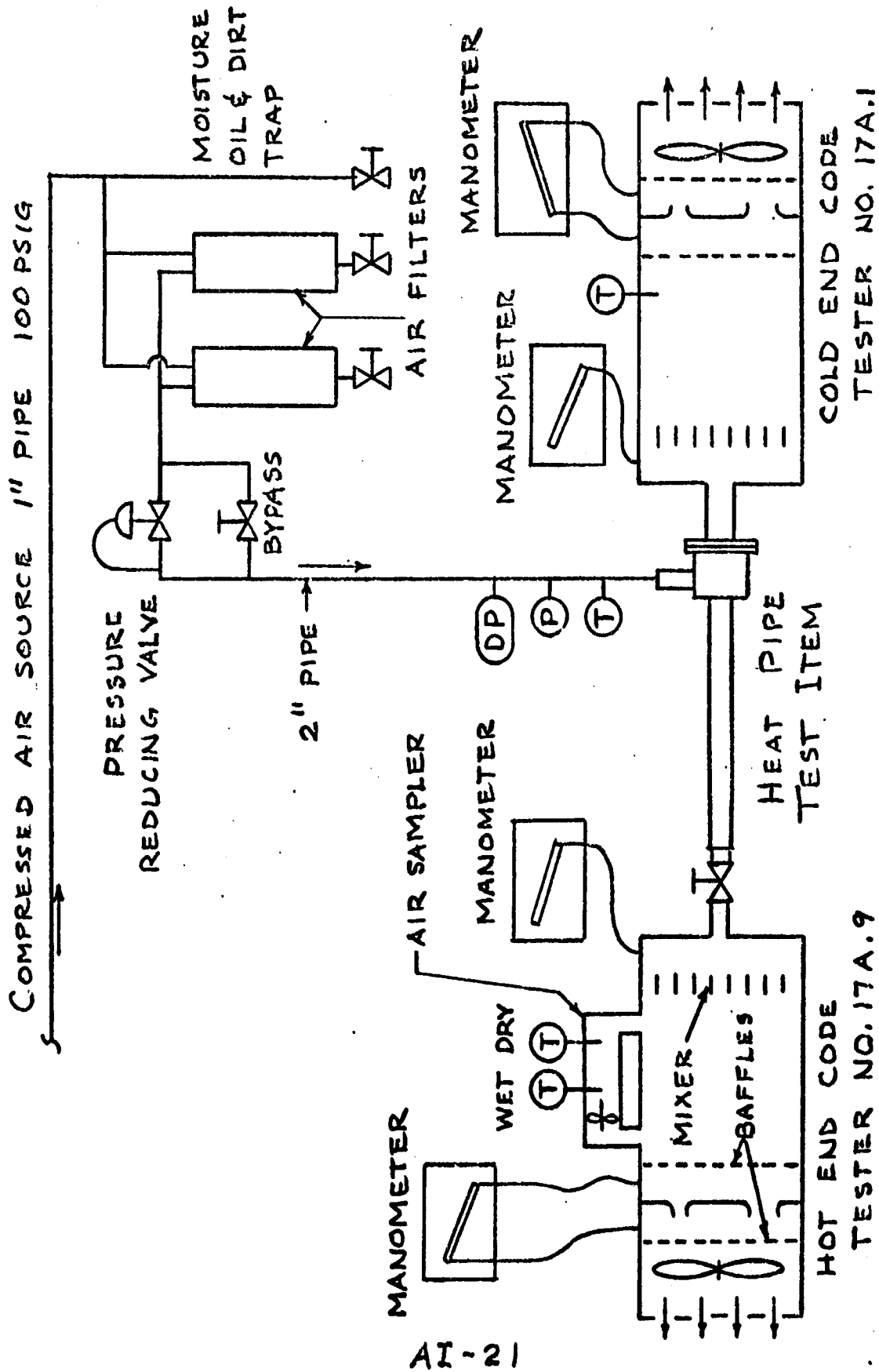


FIG. 5.

AI-20



AI-21

HEAT PIPE TEST FACILITY SCHEMATIC

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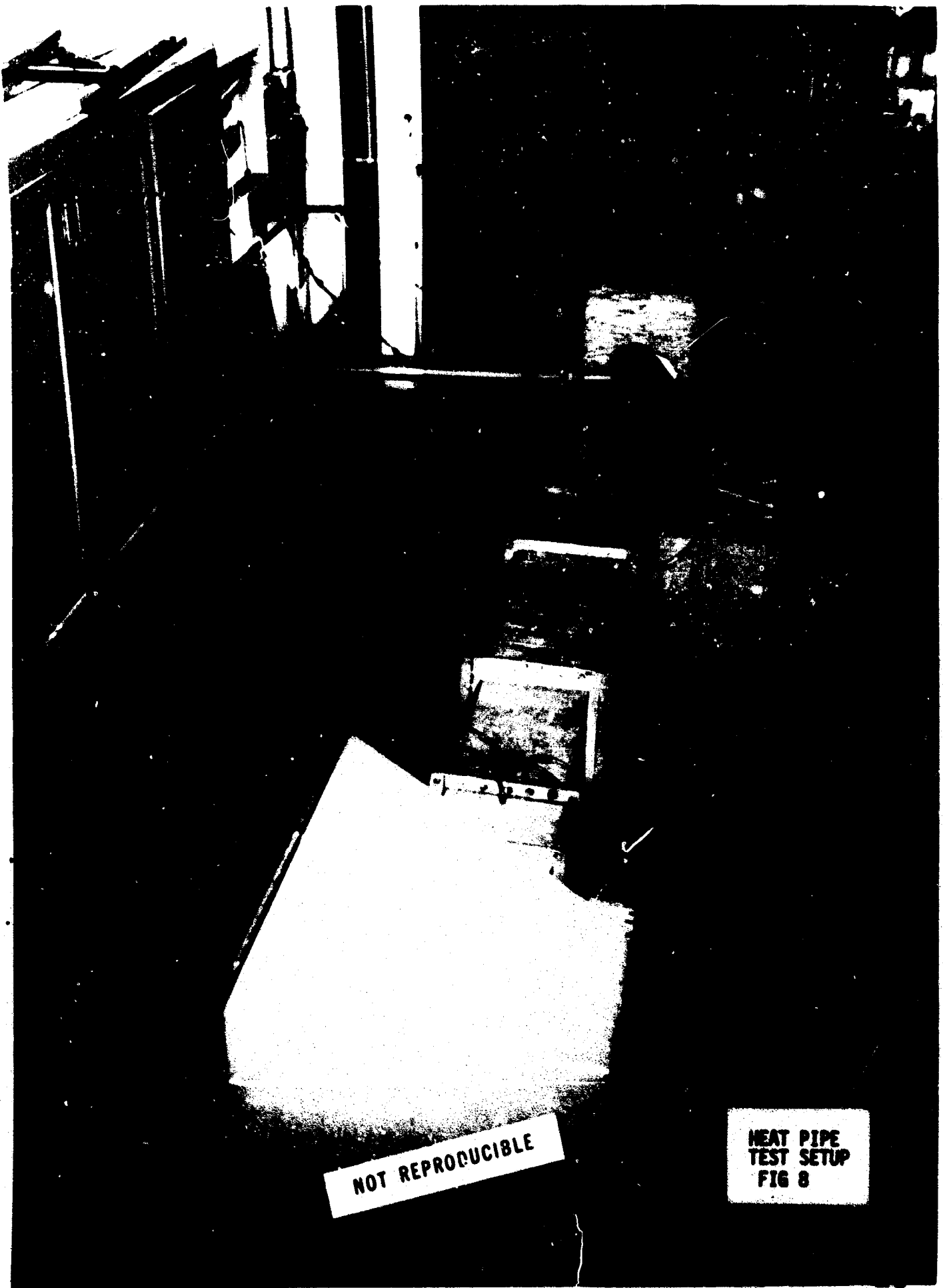
FIG. 6

SHERFY



INSULATED HEAT PIPE
FIG 7

NOT REPRODUCIBLE



NOT REPRODUCIBLE

HEAT PIPE
TEST SETUP
FIG 8

APPENDIX II

**IN-HOUSE LABORATORY INDEPENDENT RESEARCH
ANNUAL REPORT ON VORTEX TUBE COOLING**

FY 1970

BY

J. PALMER DAUPHIN

**U.S. ARMY MOBILITY EQUIPMENT
RESEARCH AND DEVELOPMENT CENTER**

Fort Belvoir, Virginia 22060

IN-HOUSE LABORATORY INDEPENDENT RESEARCH

ANNUAL REPORT ON VORTEX TUBE COOLING

June 30, 1970

Prepared By

**J. Palmer Dauphin
Project Engineer**

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NOMENCLATURE

T_1	Entering air temperature ($^{\circ}\text{R}$)
T_c	Cold air temperature ($^{\circ}\text{R}$)
T_{cs}	Isentropic cold air temperature ($^{\circ}\text{R}$)
T_h	Hot air temperature ($^{\circ}\text{R}$)
P_1	Entering pressure (psia)
P_c	Cold end pressure (psia)
k	Specific heat ratio (c_p/c_v)
m	Mass flow (lb/min)
m_c	Cold air mass flow (lb/min)
m_h	Hot air mass flow (lb/min)
JT	Joule Thomson temperature drop resulting from pressure drop (4°F for air at 100 psig to 0 psig pressure drop)
f	Relative temperature drop $(T_1 - T_c - \text{JT}) / (T_1 - T_{cs})$
u	Cold air mass-fraction (m_c/m)
v	Hot air mass-fraction (m_h/m)
Q_c	Cooling capacity (BTUH)

IN-HOUSE LABORATORY INDEPENDENT RESEARCH

ANNUAL REPORT

June 30, 1970

I. TITLE: Vortex Tube Cooling

II. PROJECT DIRECTOR: Oscar Oldberg/J. Palmer Dauphin

III. OBJECTIVE: Determine vortex tube efficiency improvement using a finned hot tube and design and fabricate a low pressure vortex tube.

IV. ACCOMPLISHMENT DURING FY 70:

A. Background. The vortex tube is a "T-shaped" device (Figure 1) with no moving parts. It separates a stream of compressed air into two streams at lower pressures, one hotter than and the other colder than the entering stream. "Hot" and "cold" air temperatures can be modulated by a change in proportioning of total air between the "hot" and "cold" ends (accomplished with a valve at the "hot" end). Figure 2 illustrates for a commercially available vortex tube, temperature performance versus inlet pressure for various proportionings of total air. Notice the "cold" air has reached its minimum temperature at 40 psig or less, and that greater percentages of "cold" air require lower inlet pressure to achieve minimum temperature. Further increases in pressure serve only to force more air through the device.

Compressed air enters the "hot" tube tangentially creating a rotating column of air (resembling a tornado) stretching the length of the "hot" end. This rotating air column is called a vortex and in our case a "forced" vortex - i.e. of constant angular velocity (ω) at all radii (r) in a given cross section - as opposed to a "free" vortex ($r\omega = \text{constant}$) where angular velocity increases as radius decreases and a constant energy level exists throughout the cross section (conservation of angular momentum). Figure 3 illustrates the velocity profile of a tornado cross-section: Peripheral velocity is small at large radii but increases in "free" vortex fashion as radius decreases until viscosity of the air causes transition to a "forced" vortex and tangential velocity then decreases with radius decrease - the air actually does work on itself causing this transition. Visualizing the vortex at a "bathtub drain" and thinking in terms of the vortex tube: If air rotates inward from a peripheral radius of one (1) unit (tangential inlet) to a drain at radius $\frac{1}{2}$ unit ("cold" tube), the natural tendency is to establish a "free" vortex with the speed at the drain being twice the initial speed; for a "forced" vortex the drain speed will be $\frac{1}{2}$ the initial value. Kinetic energy (proportional

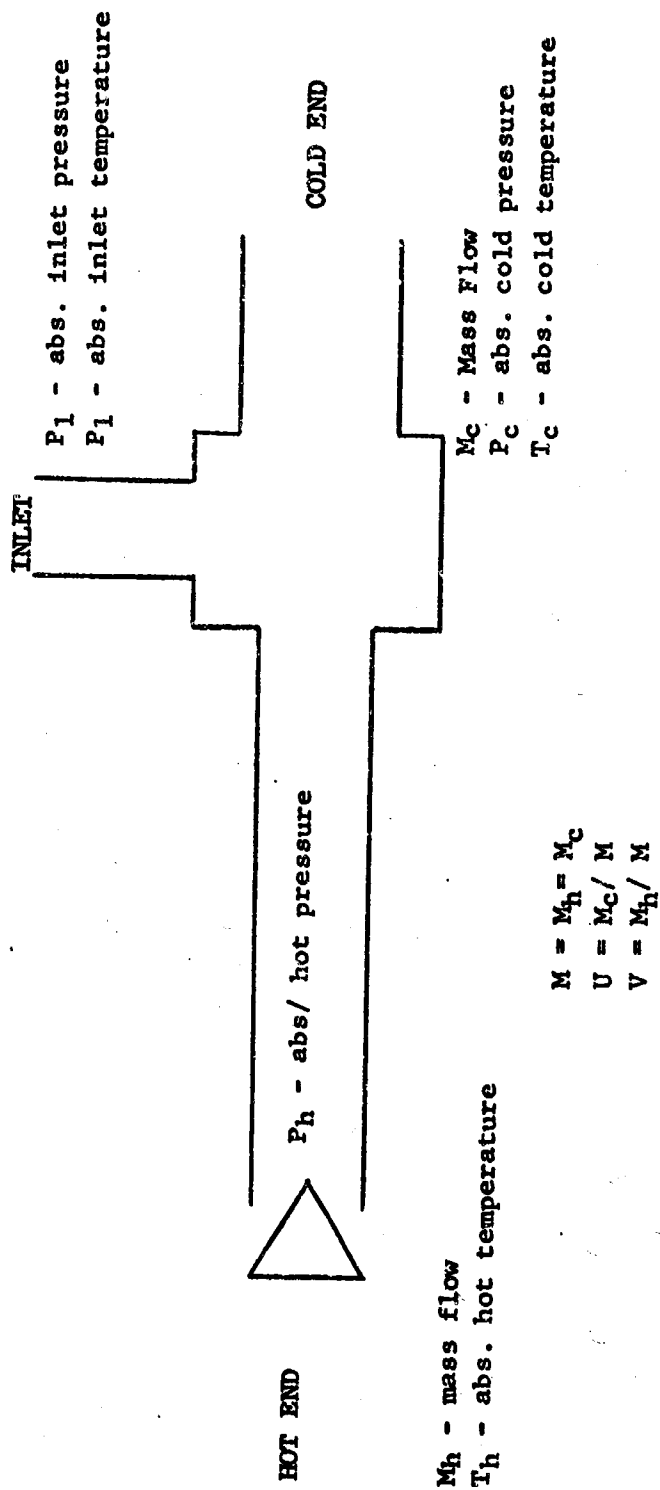


FIGURE 1
Vortex Tube Schematic

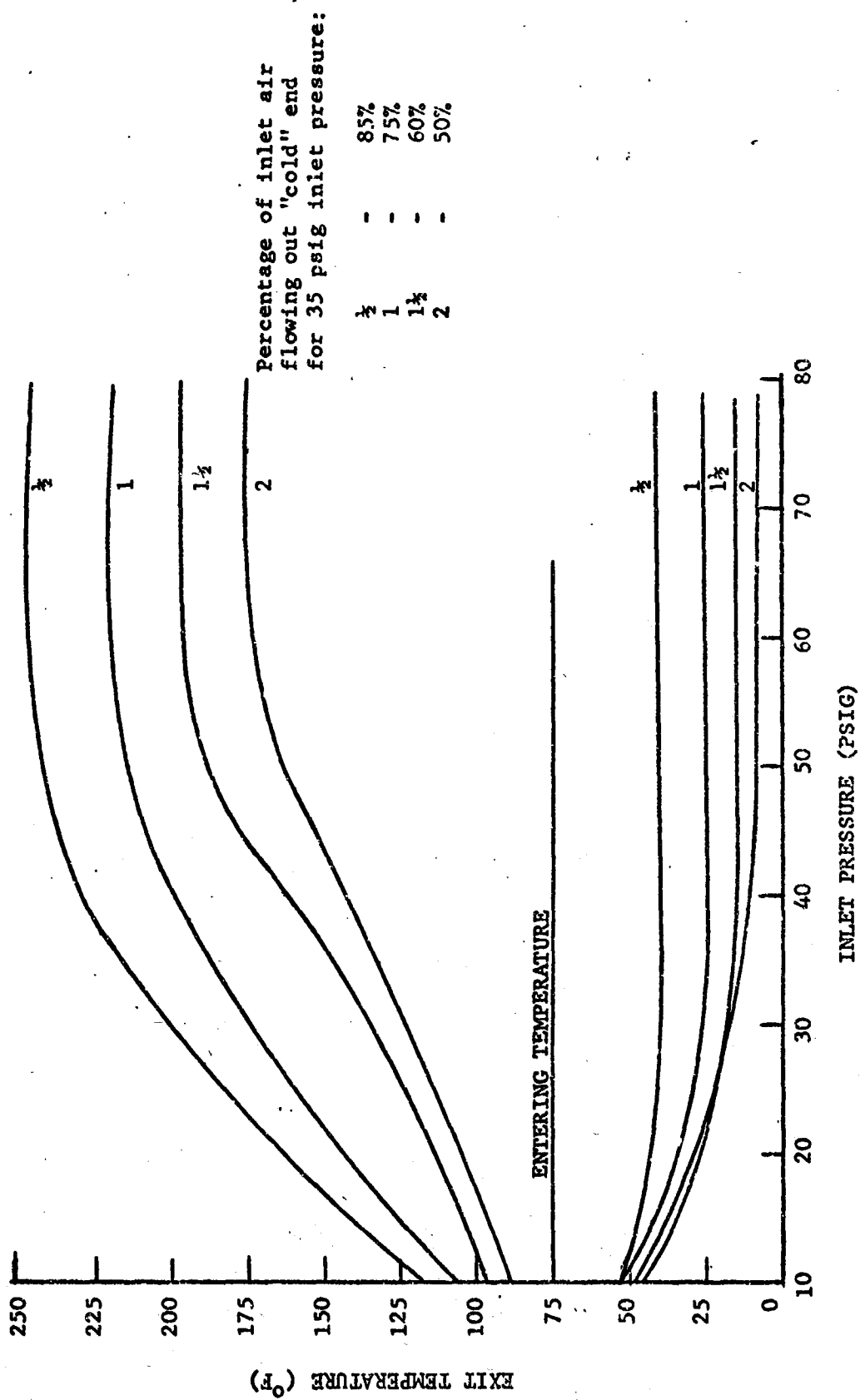


FIGURE 2
Temperature vs Inlet Pressure Performance
(Commercial 25 scfm Tube)

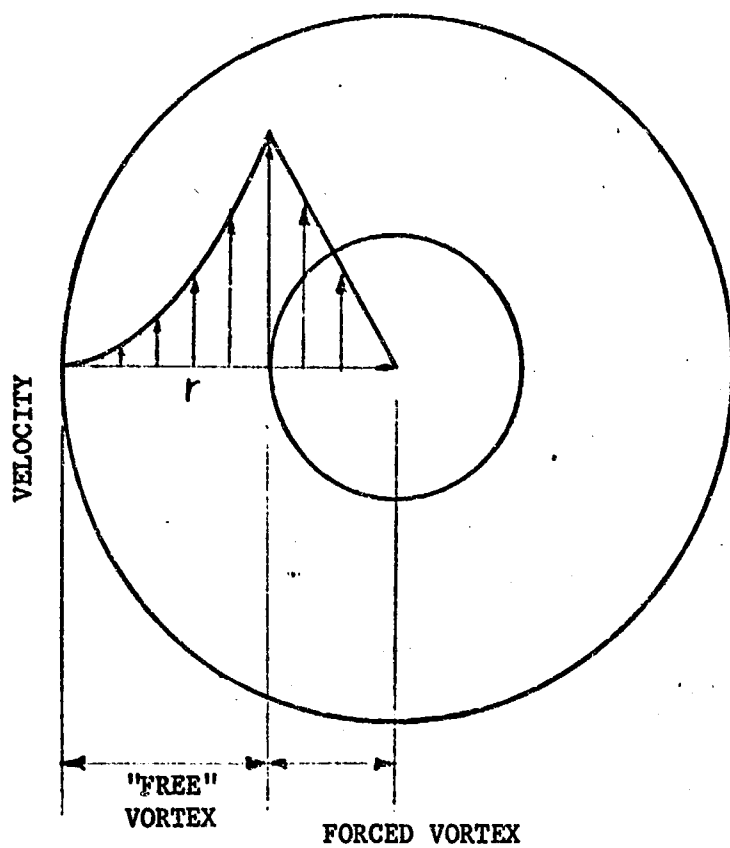


FIGURE 3
TORNADO CROSS SECTION

to square of speed) of air entering the drain will be 4 units for "free" vortex flow versus $\frac{1}{4}$ unit for "forced" vortex. Theoretically speaking then, the "cold" air in a vortex tube will transfer 15/16 of its kinetic energy to the "hot" when the "cold" tube diameter is half that of the "hot" tube.

The vortex tube is comparable to a radial-inflow turbine where compressed air expands to high velocity then transfers a portion of the velocity energy to a turbine wheel causing the wheel to rotate at high velocity. The air is cooled when it transfers velocity energy to the wheel doing work on the wheel. "Hot" air is the turbine wheel in the vortex tube. A perfect such turbine will produce an isentropic expansion producing a "cold" air temperature calculated from:

$$\frac{T_{es}}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} \quad (1)$$

The vortex tube can produce approximately 64% of this isentropic temperature drop when the cold mass fraction is near zero, an advance from 55% during the past several years. However the temperature drop decreases as more air is extracted "cold"; the reason is twofold: First, there is less "hot" air thus less resistance to rotation, resulting in higher "cold" air velocities entering the "cold" tube (less energy transfer); second, the "hot" air grows rapidly hotter increasing heat leakage into the "cold" stream. A finned "hot" tube will help offset this latter condition.

B. Finned "Hot" Tube. An aluminum "hot" tube with fins was fabricated and tested to determine effect on cooling performance. Figure 4 illustrates the modified vortex tube. Air at the temperature of that entering the vortex device was circulated over the "hot" tube at an average 550 ft/min velocity producing a 10% increase in cooling capacity for 10 psig inlet pressure.

C. Low Pressure Vortex Tube. A literary survey revealed the first book regarding the vortex. The book was published in Russia in late 1969, and is entitled, "The Vortex Effect and Its Application in Technology." A translation was performed through the MERDC Technical Library and a copy is being reviewed. The literary survey indicated no previous effort to design or determine performance of vortex tubes operating from compressed air in the zero to 30 psig range. Several investigators attempted to mathematically analyze flow and energy separation within the vortex tube, but none succeeded because of the complex equations involved.

The following design parameters were selected to provide a basis for comparison with an air-cycle system proposed for another project:

Inlet-air mass flow - 24 lb/min
Inlet-air pressure - 10 psig
Inlet-air temperature - 90°F db dry air

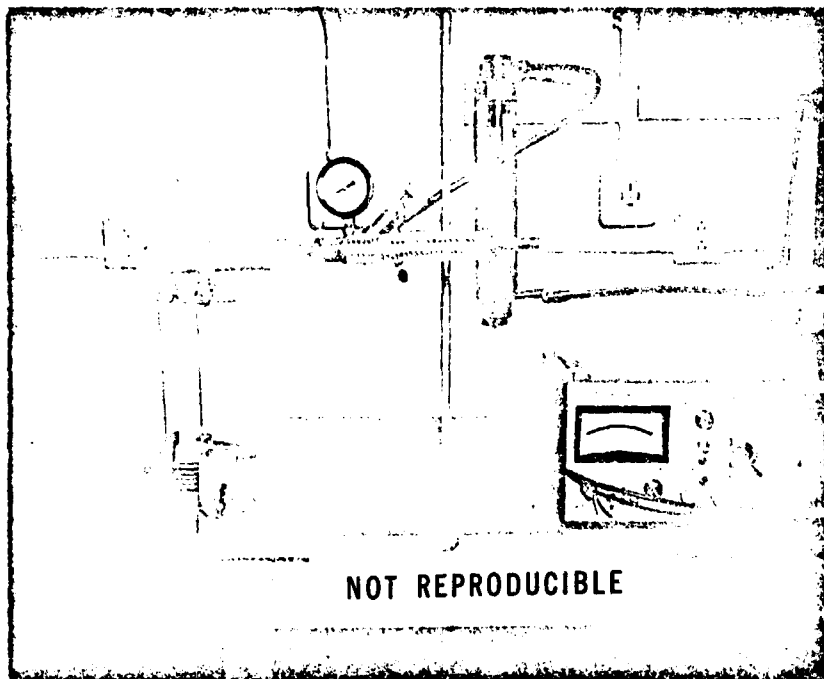


FIGURE 4
Vortex Device with Finned "Hot" Tube

Dr. Darby Fulton, Vortec Corporation (commercial manufacturers of vortex tubes), agreed to assist the design effort. Dr. Fulton is a recognized authority with over twenty years of vortex tube experience. However, he suffered a cerebral hemorrhage during the design effort leaving him unable to speak or use his left side. The decision was made to base the design on the best available commercial tube operating from 10 psig compressed air. Figure 5 illustrates temperature versus cold mass fraction performance for the commercial tube with 10 psig, 72°F inlet conditions. At 75% cold fraction the tube produces a 27°F drop, 36.7% of the isentropic temperature drop. It flows 12.5 scfm of inlet air at 10 psig (75 scfm at 100 psig). The military design requires 324 scfm at 10 psig. A diameter for the military version was determined from the square root of required scfm to actual scfm ratio (324/12.5). All remaining dimensions except for the nozzles were proportioned to the diameter. The nozzles, used to direct inlet air tangentially into the "hot" tube, were increased in number from six to 18 to reduce the head diameter. Figure 6 presents the commercial and military tubes.

Performance of proposed tube should be the same as that in Figure 5. The following equation is proposed for calculating "cold" air temperature in the 60% to 85% cold fraction range:

$$T_c = T_i - T_i \left[1 - \left(\frac{P_c}{P_i} \right)^{\frac{k-1}{k}} \right] \left[(.29 + .591u)(\sqrt{1-u}) \right] \quad (2)$$

"Cold" air temperature varies linearly with inlet temperature for constant cold fraction and cold to inlet pressure ratio:

$$\left(\frac{T_c}{T_i} \right)_1 = \left(\frac{T_c}{T_i} \right)_2 \quad (3)$$

And furthermore:

$$\frac{(T_i - T_c)_1}{(T_i - T_c)_2} = \frac{(T_i)_1}{(T_i)_2} \quad (4)$$

Tables I, II, and III indicate predicted cooling capacities calculated as follows:

$$Q_c = 1.075(324)(u)(T_i - T_c) \quad (5)$$

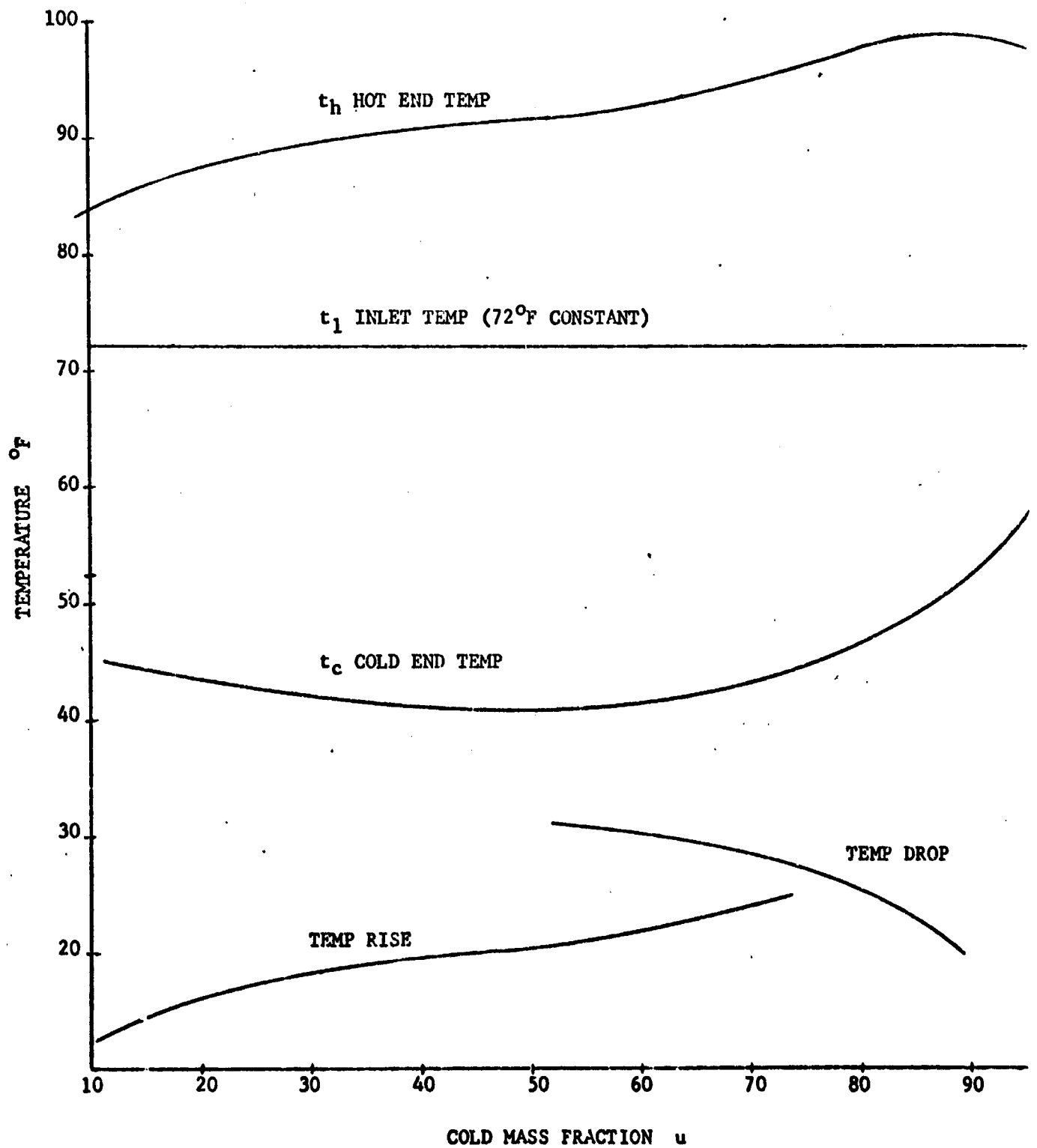
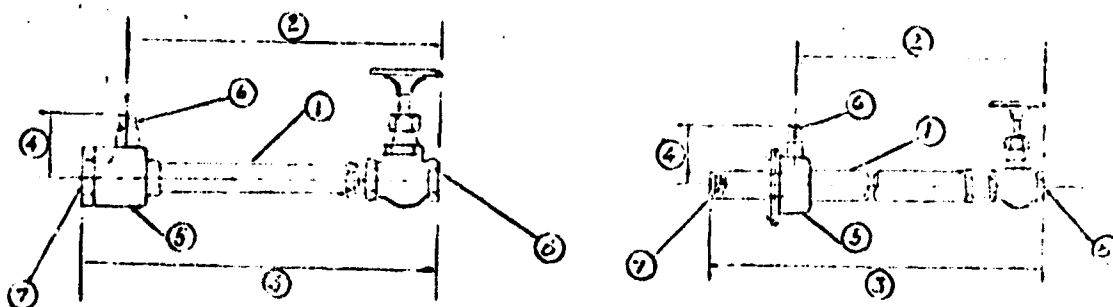


FIGURE 5
Performance for 10 psig Inlet Pressure

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<u>Item</u>	<u>Commercial</u>	<u>Military</u>
(1) "Hot" Tube	.870" I.D.	4.38" I.D.
(2) "Hot" End	11.75"	44.5"
(3) Overall	13.75"	53.12"
(4) Air Inlet	2.25"	11.75"
(5) Head	1.875" I.D.	9.5" I.D.
(6) Inlet Tube	$\frac{1}{2}$ NPT	$2\frac{1}{2}$ NPT
(7) "Cold" Tube	1 NPT	5 NPT
(8) "Hot" Valve	$\frac{3}{4}$ Globe Valve	2 Gate Valve Modified

FIGURE 6
Commercial & Military Design Vortex Tubes

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TABLE I

72°F, 10 psig inlet conditions

Cold Fraction (u)	.70	.75	.80	.85
Temp. Drop ($T_1 - T_c$)	28.2	26.9	25.1	22.5
Cooling Capacity (Q_c)	6875	7027	6994	6661

TABLE II

90°F, 10 psig inlet conditions

Cold Fraction (u)	.70	.75	.80	.85
Temp. Drop ($T_1 - T_c$)	29.2	27.8	25.9	23.3
Cooling Capacity (Q_c)	7108	7265	7230	6886

TABLE III

125°F, 10 psig inlet conditions

Cold Fraction (u)	.70	.75	.80	.85
Temp. Drop ($T_1 - T_c$)	31.0	29.6	27.6	24.7
Cooling Capacity (Q_c)	7560	7727	7691	7325

Although capacity increases with inlet temperature, "cold" air temperature increases also.

D. Discussion. A fan was used in the finned "hot" tube experiment for blowing air over the fins, whereas a jet syphon would be used in a finalized design. The jet syphon would utilize the "hot" air velocity to induce airflow over the fins and increase capacity for no additional operating cost. Also, a recuperator between inlet air and "cold" air should increase performance at no addition to operating cost.

E. Personnel. Oscar Oldberg, MERDC, performed the finned hot tube experiment. Palmer Dauphin, MERDC, with assistance from Dr. Darby Fulton, Vortec Corporation, performed the literary survey and military tube design.

V. FUTURE PLANS: Original plans to fabricate and test the proposed military design have changed, largely due to cost. Instead, three commercial tubes of various sizes were purchased for the equivalent in-house fabrication cost for one military design. The largest of the three tubes approaches the proposed military tube. Future plans now involve testing the three tubes and deriving design equations.

VI. FISCAL STATUS:

a. Funds allocated in FY 70	\$8,270.00
b. Funds required in FY 71	4,000.00
c. Previous year funds available	0

APPENDIX III

VORTEX TUBE HEATING/COOLING

**Excerpted from USAMERDC Pamphlet, subject as above,
dated 25 March 1969, from Environmental Equipment
Division**

APPENDIX III

Vortex Tube Heating/Cooling

The vortex tube, a device with no moving parts, emits cold air from one end and hot air from the other end when connected to a source of compressed air.

It was invented in the early 1930's by MG Ranque, however little was known about it until the mid 1940's when rudolph Hilsch published his studies on the device. It has also been called the Ranque tube, Hilsch tube, Ranque-Hilsch tube, vortex generator, T-tube, or Separator tube.

The two basic types of vortex tubes are the direct flow and counterflow. Both types have an inlet nozzle(s), a "hot" tube, "cold" tube, and throttling valve; in addition, the counter flow type has an orifice plate.

The objective in all types of vortex tube is to establish a rotating mass of air within the hot tube. This accomplished by locating the inlet nozzle(s) perpendicular and tangent to the "hot" tube. As the compressed air enters the "hot" tube from the nozzle, it expands and moves toward the center in a screw-like fashion. The air speeds-up as it moves toward the center, much like an ice skater spins faster as he pulls his hands in toward his body.

We now have a rotating mass of air which is in essence, a small tornado where the velocity of the air is high at the center and slow at the periphery. Due to the viscosity (internal friction) of the air, the faster air in the center tries to speed-up the slower air at the periphery; thus the higher velocity air transmits part of its energy to the slower air. In other words, the air in the center does work on the air at the periphery. Since temperature is a measure of internal energy, the center air is now colder and the peripheral air hotter than the air that entered through the nozzle. By partly closing the throttling valve at the exit of the "hot" tube we create a pressure in that tube and thus force the cold air in the center out the "cold" tube.

The temperature of both ends can be modulated by varying the proportioning of total air between the two ends. This would allow the use of a fluidic thermostat and eliminate electrical controls.

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